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Using cryogenic exergy of liquefied natural gas for electricity production with the Stirling cycle

Hui Dong, Liang Zhao*, Songyuan Zhang, Aihua Wang, Jiuju Cai

SEP Key Laboratory of Eco-Industry of Northeastern University, Shenyang 110819, China

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ABSTRACT

Cryogenic generation is one of the most important ways to utilize cold energy during LNG (liquefied natural gas) regasification. This paper fundamentally investigates LNG cryogenic generation with the Stirling cycle method based on previous studies. A basic process of LNG cryogenic generation with the Stirling cycle was presented initially with seawater and LNG as heat source and heat sink. And its thermodynamic analysis was performed to verify the theoretical feasibility of the Stirling cycle method. The generating capacity, the exergy efficiency and the cold energy utilization efficiency of the basic process were also calculated. Subsequently, the influences of evaporation pressure on net work, equipment performance and comprehensive efficiency of cold energy utilization were discussed and the effect of LNG mass flow as well as the ambient temperature was also studied. Finally an improved process of LNG cryogenic generation with Stirling cycle method combined with an air liquefaction process is proposed as feasibility in improvements of the basic process.

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1. Introduction

LNG (Liquefied natural gas) has a large temperature difference (about 182 K) between ambient air and water, and thus contains considerable cold energy which can be recovered for electricity generation, air separation, food freezing etc. Using cold energy to generate electricity is more efficient than using waste heat with the same temperature difference [1]. Thus, cryogenic generation is the most effective way to recover the cold energy of LNG. What's more, many previous studies about more effective way to use LNG cold energy for power generation have been conducted. Wang et al. [2] proposed an ammonia–water power system with LNG as its heat sink to utilize the low grade waste heat, and optimized the key thermodynamic design parameters. Szargut et al. [3] investigated three variants of cryogenic power plant and studied the influence of the changing ambient temperature on the efficiency of the cryogenic power plant. Dispenza et al. [4,5] proposed an innovative process which uses LNG as the cold source in an improved CHP (combined heat and power) plant, and analyzed the performance based on thermodynamics and economics. Meanwhile, to improve the cold energy recovery efficiency of an LNG cryogenic power plant, some studies have been carried out. Tsatsaronis and Morosuk [6,7] presented a detailed advanced exergetic analysis of a novel co-generation concept that combines LNG regasification with the generation of electricity. In recent years, researches for the production of electricity form LNG by utilizing its cryogenic energy have developed in China. Liu and Guo [8] proposed a novel cryogenic cycle by using a binary mixture as working fluids and combined with a vapor absorption process for LNG cold energy recovery. Lu et al. [9] proposed a cascading power cycle with LNG directly expanding consisting of a Rankine cycle with ammonia–water as working fluid and a power cycle of combustion gas to recover cold energy of LNG.

Japan is the most successful country in the cryogenic generation application field. It has built around 15 cryogenic power plants from 1979 to 2000 [10]. These plants operate by using the Rankine cycle method or NG (natural gas) directly expanding method or combination type of these two methods. However, the recovery efficiency of LNG cold energy in Japan is usually about 14% [11]. No matter in academic research field or in engineering application field, ORC (organic Rankine cycle) is the most commonly used to generate electricity for LNG cold energy recovery. Compared with ORC, ideal Stirling cycle has several advantages in theory, such as higher thermal efficiency, larger cycle net work etc. [12]. Moreover, the device works with Stirling cycle – Stirling engine, has many merits like operation with low noise, operation with constant power output, consist of fewer parts, etc. These advantages give Stirling cycle engine vast application potential.

In 1978, a Japanese researcher named Oshima [13] proposed a scheme using LNG or LH_2 (liquid hydrogen) cold energy for power generation by a cryogenic type Stirling engine. Many years later, Servis

* Corresponding author. Fax: +86 024 83672218.
E-mail addresses: zhliang8341@163.com, 463567011@qq.com (L. Zhao).

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[14] innovatively proposed the NSCE (New Stirling Engine Concept) to develop a large cryogenic type Stirling engine by adding a isobaric heat addition process in the cycle, which has already authorized in several patents [15,16]. The application possibilities of using the NSCE for generating power coupled with LNG regasification process were also investigated by Servis [17]. Meanwhile, Tan [18] made a preliminary discussion that Stirling engine is evidently more favorable to recover LNG cold energy. Li [19] put forward a complex system comprising a gas turbine along with a Stirling engine to utilize LNG cold energy and thermodynamically analyzed the system. However, these preliminary researches do not explain the principle of LNG cryogenic generation with Stirling cycle method in depth.

Based on the existing researches, we propose a basic Stirling cycle cryogenic generation process for LNG cold energy recovery in this paper. The energy and exergy changes in thermodynamic process, both of LNG regasification process and Stirling cycle operation process are calculated. The effects of some key parameters on performance of the basic process, such as LNG vaporization pressure and ambient temperature, are also investigated. Additionally, we put forward an improved process of LNG cryogenic generation with Stirling cycle combined with air liquefaction to study the feasibility for optimizing the basic process.

2. Basic process description

We propose a basic Stirling cycle cryogenic generation process. The scheme of the basic Stirling cycle cryogenic generation process for LNG cold energy recovery and the corresponding *T*–*s* diagram are shown in Figs. 1 and 2, respectively. The basic process includes two branches: a LNG regasification process and a nitrogen Stirling cycle process. ①The LNG regasification process is shown as 1L–2L–3L–4L–5L. From the beginning of the basic process, LNG is stored at −162 °C under 0.14 MPa. And then, LNG is pumped into the pipeline from a storage tank and pressurized to vaporization pressure (from state 1L to 2L). It is then heated to saturated temperature in heat exchanger HX3 (heat exchanger 3) (2L–3L). Afterwards, it flows through the nitrogen compressor to cool down the nitrogen and is partly gasified (3L–4L). Finally, LNG is totally gasified through the heat exchanger HX2 and heated to the supply temperature (4L–5L). ②The nitrogen Stirling cycle process is depicted as 1–2–3–4–1. Nitrogen goes into the seawater-heating turbine, and isothermally expands from state 1 to state 2. Then it flows into heat regenerator (HX1) and gives off heat isochorically (state 2 to state 3). Subsequently, nitrogen is isothermally compressed by a LNG-cooling compressor (state 3 to state 4). Eventually, it goes through the heat regenerator (HX1) again and takes heat back (to state 1). Thus, nitrogen completes an ideal Stirling cycle.

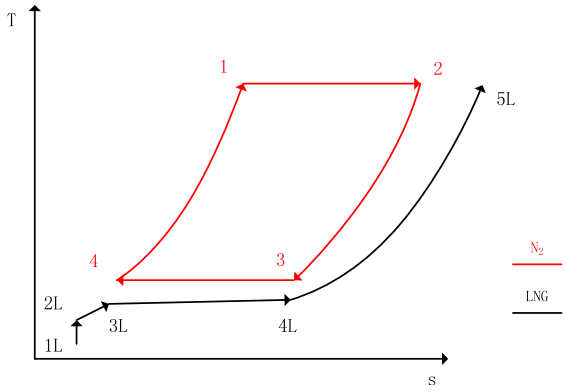

Fig. 2. *T*–*s* diagram of the basic cycle.

LNG is a mixture (consists of methane, ethane, propane, etc.) without a fixed phase-transition temperature at a constant pressure. However, it may exist in a relatively stable phase-transition period with little temperature change. We call this relatively stable phase-transition period the gasification level section. The Stirling cycle requires constant-temperature heat source and heat sink for minimizing irreversible loss. Therefore, when LNG is regarded as the heat sink of the Stirling cycle for power generation, the isothermal compression process (state 3–4), should be in the gasification level section.

In this article, the isothermal compression process and the isothermal expansion process of Stirling cycle are respectively implemented by a constant temperature compressor and a constant temperature turbine. The compressor is cooled by LNG and isothermally compresses nitrogen. The turbine is heated by seawater and drives a generator to produce electricity. On the other hand, nitrogen is chosen as the working fluid in the ideal Stirling cycle in this paper instead of hydrogen and helium, for it is substantial and easy to produce.

3. Basic process performance

3.1. Parameters calculation

Assumptions:

(1) LNG is a mixture that consists of methane, ethane, propane and nitrogen, which is listed in Table 1.

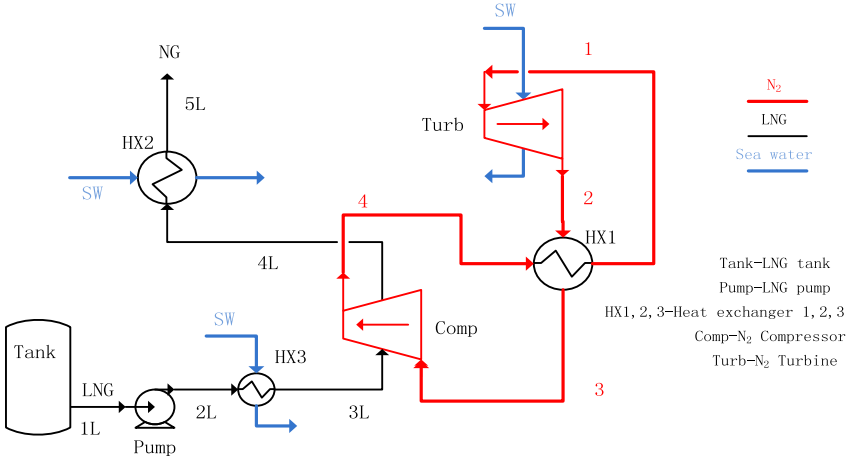

Fig. 1. The detailed process of the basic cycle.

Table 1
Mole percentages of LNG components.

Component	CH_4	C_2H_6	C_3H_8	N_2
%	90.38	5.37	4.04	0.21

(2) The gasification flow rate is 1 kg/s.
(3) The isothermal compression and isothermal expansion process of the Stirling cycle may be achieved.
(4) According to reference [20], the pressure loss of LNG vaporization system could be 3%. Thus, the influence of pressure loss in this paper may be neglected in calculation process.
(5) Considering the safety of environment, the temperature of seawater output from this basic process may be close to ambient temperature. Accordingly, for all heat exchangers involving seawater, the seawater inlet and outlet temperatures seawater are set to be 25 °C and 15 °C, respectively.
(6) The heat sink temperature of the Stirling cycle should be 15 °C higher than the saturated liquid temperature under its corresponding pressure.
(7) At a constant vaporization pressure, the gasification level section starts from the LNG saturated liquid point and ends when the LNG temperature increase 10 °C.

Among many state equations, the P–R (Peng–Robinson) equation is suitable for calculating the physical properties of such mixtures as LNG [21]. Therefore, we chose P–R equation [22] to calculate the properties of LNG as well as nitrogen for its simple form and acceptable accuracy.

The LNG and nitrogen properties shown in Tables 2 and 3 are calculated by the P–R equation.

3.2. Thermodynamic calculation

To simplify calculation, pressure loss in pipelines and heat exchangers are ignored. During the LNG regasification process, pressure is kept constant from state 2L to 5L. The heat input into system may be expressed by Eq. (1). The corresponding results are shown in Table 4.

$$q = \Delta h_{\mathrm{LNG}} \tag{1}$$

The nitrogen Stirling cycle involves two isothermal and two isochoric processes. Heat and work changes calculated by Eqs. (2)–(5) are listed in Table 5.

For isothermal process,

$$q_T = T \cdot \Delta s_{\mathrm{N_2}} \tag{2}$$

$$w_T = q_T - \Delta h_{\mathrm{N_2}} \tag{3}$$

For isochoric process,

$$q_v = \Delta u_{\mathrm{N_2}} \tag{4}$$

$$w_v = 0 \tag{5}$$

Table 2
The state parameters of LNG.

Fluid states	t (°C)	p (MPa)	h (kJ/kg)	s (kJ/kg K)
0	20	0.1	−4269	10.35
1L	−162.0	0.14	−5144	4.189
2L	−160.5	3.0	−5136	4.213
3L	−92.21	3.0	−4892	5.882
4L	−82.30	3.0	−4664	7.119
5L	5	3.0	−4342	8.565

Table 3
The state parameters of nitrogen.

Fluid states	t (°C)	p (MPa)	h (kJ/kg)	s (kJ/kg K)	v (m^3/kg)
0	20	0.1	−5.482	5.272	0.8696
1	10	4.280	−27.767	4.085	0.0193
2	10	0.378	−16.727	4.839	0.2218
3	−77.3	0.260	−106.948	4.568	0.2218
4	−77.3	2.780	−121.922	3.810	0.0193

The heat quantity absorbed by LNG in process 3L–4L is equal to the heat quantity released by nitrogen in process 3–4. And then, the nitrogen mass flow rate is calculated as 1.54 kg/s. The net work of the basic process is 100.08 kW (M_{LNG} = 1 kg/s). The seawater flow rates in heat exchanger HX2 and HX3 are 7.45 kg/s and 5.66 kg/s, respectively. By calculation, when LNG flow rate is raised to 60 t/h (16.67 kg/s) (vaporization pressure is also 3.0 MPa), the net work output of the basic process becomes 1667 kW.

Here, the assumptions for the Stirling cycle such as isothermal compression, isothermal expansion and no pressure loss, etc. may account for the resulting output power increase and overestimate the enhancement. Considering practical conditions, the output power should be lower than 1667 kW. For example, according to the test results of NSC (PROBA3D engine) in Ref. [14], when working under the same conditions, the power output of the basic process with PROBA3D engine is about 1389 kW and the engine efficiency is around 50%. Another example is that, compared with the cryogenic power station in Senboku Daini (Japan) [10], the power output increases 14.97% in case of working with the basic process, but it decreases around 4% when the practical conditions are considered. On the other hand, we believe that the developing of Stirling engine technology will boost the out power of the basic process.

The cryogenic power coefficient of the basic process can be defined as the ratio of electricity generating capacity to the consumption of the cold energy as expressed below.

$$x = \frac{W_g}{Q_c} = \frac{\eta_g W_{\mathrm{net}}}{Q_{\mathrm{3L-4L}}} \tag{6}$$

where: W_g, Q_c is power output of basic process and cold energy into the basic process respectively. η_g is the machinery conversion efficiency.

The cold energy used for the Stirling cycle comes from the LNG regasification process (3L–4L). Supposing that the machinery conversion efficiency between the Stirling engine and the generator is 90%, the cryogenic power coefficient will be 39.42% (M_{LNG} = 1 kg/s, p_{LNG} = 3.0 MPa).

3.3. Exergy analysis of basic process

Fig. 3 shows the exergetic analysis schematic of the cycle. The reference temperature and pressure are respectively at 20 °C and

Table 4
The heat needed in LNG gasification process.

Processes	2L–3L	3L–4L	4L–5L
q (kJ/kg_{LNG})	244.13	228.04	321.43

Table 5
Heat and work changes of nitrogen.

Process	1–2	2–3	3–4	4–1
q_T (kJ/kg_{N2})	213.4951	−64.82	−148.5032	64.78
w_T (kJ/kg_{N2})	200.4551	0	−135.5309	0

under 0.1 MPa (the reference values of entropy and enthalpy are shown in Tables 2 and 3). The physical exergy $ex_{T,p}$ of LNG and nitrogen can be calculated through Eq. (7). Their calculating results are listed in Table 6.

$$ex_{T,p} = \Delta h_{LNG(N_2)} - T_0 \Delta s_{LNG(N_2)} + R_g T_0 \ln \frac{p}{p_0} \quad (7)$$

According to Fig. 3, an equilibrium equation of physical exergy can be formulated as described below.

$$\sum I = \left(\sum Ex_{SW,in} + Ex_{L1} + W_{Pump}\right) - \left(\sum Ex_{SW,out} + Ex_{L5} + W_{net}\right) \quad (8)$$

where,

$$\sum I = I_{HX1} + I_{HX2} + I_{HX3} + I_{Turb} + I_{Comp} + I_{Pump}$$

The exergy efficiency of the system in Fig. 3 is shown as follows:

$$\eta_{ex} = \frac{Ex_{gain}}{Ex_{pay}} \quad (9)$$

where,

$$Ex_{pay} = Ex_{L1} - Ex_{L5} + W_{Pump}, \quad Ex_{gain} = \sum Ex_{SW,out} - \sum Ex_{SW,in} + W_{net}$$

The exergy loss and the exergy efficiency of all equipments in the basic process are calculated based on the Eqs. (8) and (9). The corresponding results are respectively shown in Tables 7 and 8.

As shown in Tables 7 and 8, the heat exchanger HX3 has the largest exergy loss in the basic process. In other words, the exergy efficiency of the heat exchanger HX3 is the lowest in the basic process due to the large heat transfer difference, only 1.65%.

Figs. 4 and 5 respectively represent how the outlet temperature difference varies with the heat flow rate of the heat exchangers HX2 and HX3. LNG is directly heated by seawater through HX2 and HX3. The LMTD (log mean temperature difference) of HX2 and HX3 is calculated as 48.90 °C and 145.20 °C.

Table 6
The results of exergy in process.

Fluid states	LNG (kJ/kg)	Fluid states	N_2 (kJ/kg)
ex_{1L}	963.2135	ex_1	332.8484
ex_{2L}	974.9227	ex_2	118.0733
ex_{3L}	721.7027	ex_3	108.3823
ex_{4L}	580.7018	ex_4	320.3499
ex_{5L}	471.2293	–	–

Large heat transfer temperature difference results large irreversible loss and low exergy efficiencies of these two heat exchangers.

4. Parameter sensitivity analysis and discussion

The LNG vaporization pressure is a very sensitive parameter in the release process of LNG cryogenic exergy. The effects of vaporization pressure on other parameters of the basic process, such as the output power, the exergy efficiency, the cryogenic power efficiency and the comprehensive efficiency of cold energy, are mainly analyzed in this section.

4.1. On the work output of the basic process

The Stirling cycle here is designed to utilize the cold energy from the gasification level section as discussed earlier. The heat sink temperature is set to be 15 °C higher than the saturated liquid temperature. However, the saturated liquid temperature and saturated vapor temperature are influenced by the vaporization pressure.

As seen from Fig. 6, both the saturated liquid temperature and saturated vapor temperature rise with the increase of the vaporization pressure. The saturated liquid temperature shows a significant rise trend before 1.0 MPa and then a linear growth mode. Obviously, the change trend of the saturated vapor temperature is smoother than that of the saturated liquid temperature, and remains nearly unchanged when the vaporization pressure is over 4.0 MPa. In addition, the temperature difference between saturated liquid and saturated vapor decreases along with the rise of vaporization pressure.

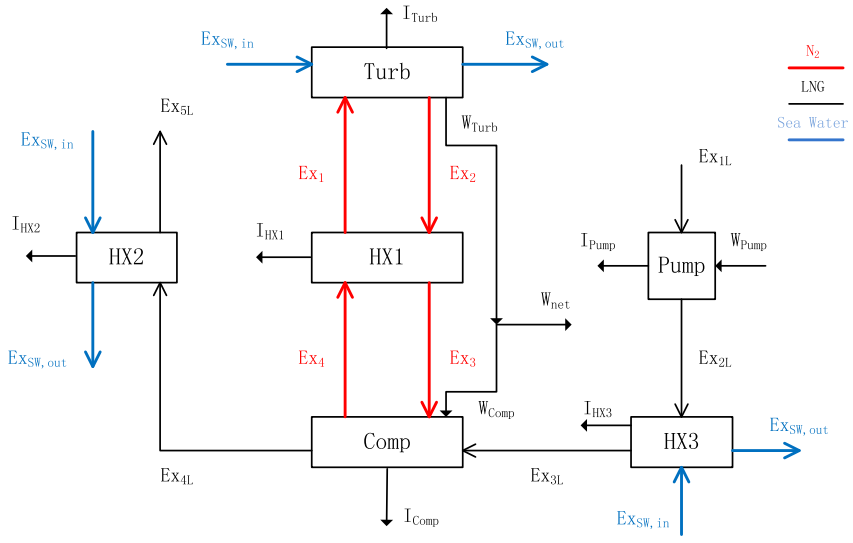

Fig. 3. Exergetic schematic of cycle.

Table 7
Exergy loss in equipments of system (M_{LNG} = 1 kg/s, p_{LNG} = 3.0 MPa).

Components	Ex_{in} (kW)	Ex_{out} (kW)	I (kW)
HX1	677.7923	673.4796	4.3127
HX2	580.7018	476.7432	103.9587
HX3	974.9227	725.8904	249.0323
Turb	511.3018	498.7702	12.5316
Comp	1093.2371	1072.8042	20.4329
Pump	974.2444	974.9227	−0.6783
System	974.2444	593.2800	380.9644

The heat absorption of the Stirling cycle from the heat source is equal to the heat of the nitrogen absorbing from the seawater, as shown in Eq. (10).

$$Q_{in} = m_{N_2} T_H \Delta s_{1-2} = m_{SW} \Delta h_{SW} \tag{10}$$

The heat release from Stirling cycle to the heat sink is equivalent to the heat absorption of LNG from nitrogen, as shown in Eq. (11).

$$Q_{out} = m_{N_2} T_L \Delta s_{3-4} = m_{LNG} \Delta h_{LNG} \tag{11}$$

Eq. (12) can be deducted from Eqs. (10) and (11), for $|\Delta s_{1-2}| = |.\Delta s_{3-4}|$. in ideal Stirling cycle,

$$\frac{Q_{in}}{Q_{out}} = \frac{T_H}{T_L} = \frac{m_{SW}\Delta h_{SW}}{m_{LNG}\Delta h_{LNG}} \tag{12}$$

The relationship (shown in Fig. 7) between the vaporization pressure and the net work of the basic process may be determined through Eq. (13) as shown below.

$$W_{net} = \left(\frac{T_H}{T_L} - 1\right) m_{LNG} \Delta h_{LNG} \tag{13}$$

Eq. (13) shows that the net work of the basic process depends on the temperature of heat source and heat sink of Stirling cycle, and also on the heat absorption capacity of LNG gasification, but is irrelevant to the working fluid of Stirling cycle. Because the heat sink temperature of the basic process and the heat absorption capacity of LNG gasification are influenced by the vaporization pressure, the net work can be expressed by the function of heat source temperature, vaporization pressure and LNG flow rate, namely, $W_{net} = f(T_H, p, m_{LNG})$.

As can be seen from Fig. 7, the net work sharply declines as the vaporization pressure rises before reaching 1.0 MPa and the decline becomes much smoother when the vaporization pressure exceeds 1.0 MPa. Besides, Fig. 8 qualitatively describes the relationship between the net work and the vaporization pressure from the view of temperature and entropy. The net work of the basic process described with the area of 1–2–3–4–1 becomes increasingly smaller as the vaporization pressure increases.

Table 8
Exergy efficiency in equipments of system (M_{LNG} = 1 kg/s, p_{LNG} = 3.0 MPa).

Components	Ex_{pay} (kW)	Ex_{gain} (kW)	η_{ex}%
HX1	19.1994	14.8867	77.54
HX2	109.4725	5.5139	5.04
HX3	253.2201	4.1878	1.65
Turb	329.9247	317.3930	96.20
Comp	346.0448	325.6119	94.10
Pump	11.7092	11.0308	94.21
System	503.0151	122.0507	24.26

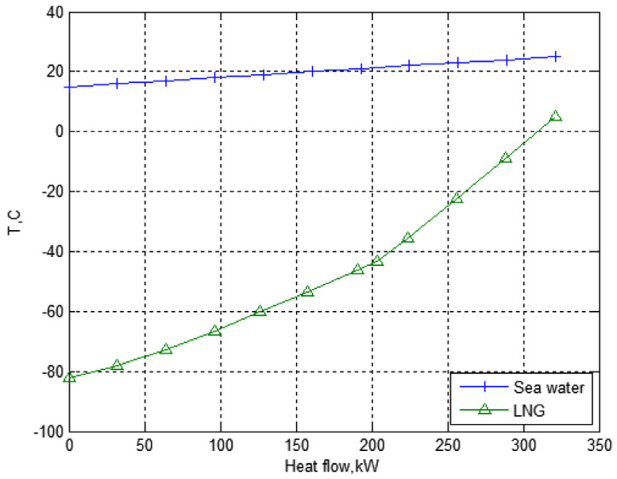

Fig. 4. The relationship between outlet temperature difference and heat flow rate in heat exchanger HX2.

4.2. On the equipment performance

4.2.1. Heat exchanger HX2

The heat exchanger HX2 is used to completely gasify LNG and achieve the supply temperature (5 °C). As the vaporization pressure increases, the initial temperature of LNG also rises before entering heat exchanger HX2. Consequently, the heat transfer temperature difference decreases. The relationship between the logarithmic mean temperature difference and the vaporization pressure in heat exchanger HX2 is shown in Fig. 9. Evidently, when the vaporization pressure below 2.0 MPa, the heat transfer temperature difference drops quickly. But it presents a much smoother decrease as the vaporization pressure exceeds 2.0 MPa.

Fig. 10 illustrates the variation of exergy loss and the thermal load along with the rise of vaporization pressure in HX2. When the vaporization pressure increases, the thermal load and the irreversible loss simultaneously decrease.

4.2.2. Heat exchanger HX3

LNG is heated in HX3 and becomes to saturated liquid before flowing into the heat sink side of the Stirling engine. Theoretically, when the isentropic efficiency of blower pump is 100%, the LNG

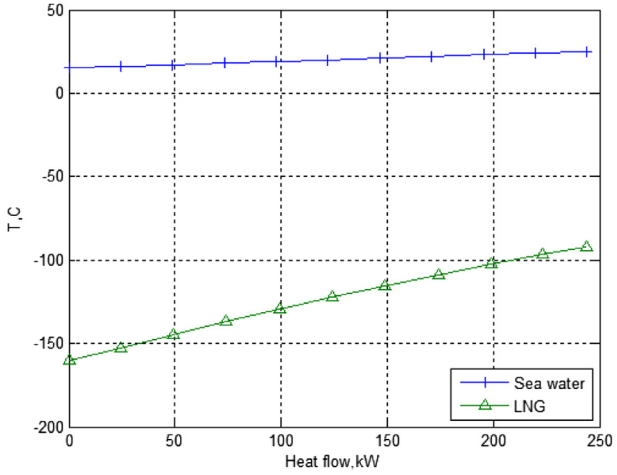

Fig. 5. The relationship between outlet temperature difference and heat flow rate in heat exchanger HX3.

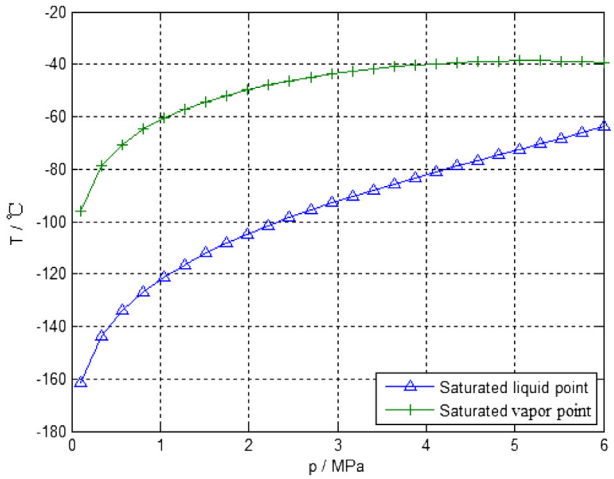

Fig. 6. The saturated liquid and saturated vapor temperature curve under the influence of vaporization pressure.

inlet temperature of HX3 just rises by 1.2 °C with the increase of LNG pressure from 0.14 MPa to 6.0 MPa. Thus, the increasing vaporization pressure has a little influence on the maximum heat transfer temperature difference of heat exchanger HX3. The relationship between the logarithmic mean temperature difference and the vaporization pressure in HX3 is also shown in Fig. 9. Its changing trend is similar to that of HX2.

The impacts of vaporization pressure on thermal load and exergy loss of HX3 are shown in Fig. 11. As the vaporization pressure increases, the thermal load and the exergy loss both rise. A cross point corresponding the pressure 3.36 MPa can be seen between the thermal load curve and the exergy loss curve. The exergy loss is greater than the thermal load below the cross point and it shows a reverse trend above the cross point. The reason is given as follows.

If there was a Carnot engine working at the same temperature conditions with HX3, this Carnot engine would be,

$$\frac{W_{\text{net}}}{Q_{\text{out}}} = \frac{\overline{T_H}}{\overline{T_L}} - 1 \tag{14}$$

where W_{net} is the work production of the Carnot engine, Q_{out} is the heat released from the Carnot engine. $\overline{T_H}$ and $\overline{T_L}$ are the mean endothermic and mean exothermic temperatures, respectively.

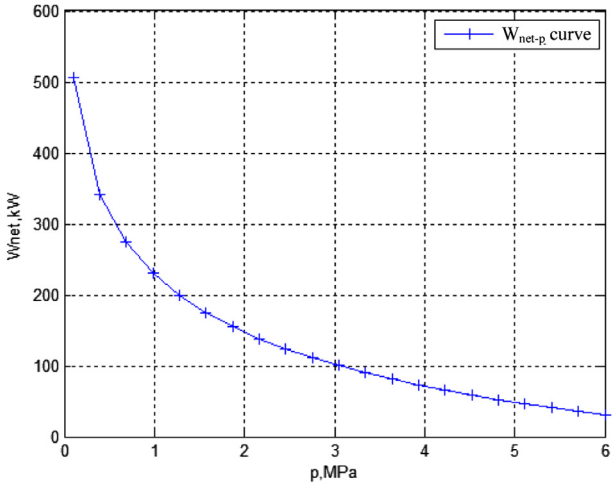

Fig. 7. The relationship between vaporization pressure and net work.

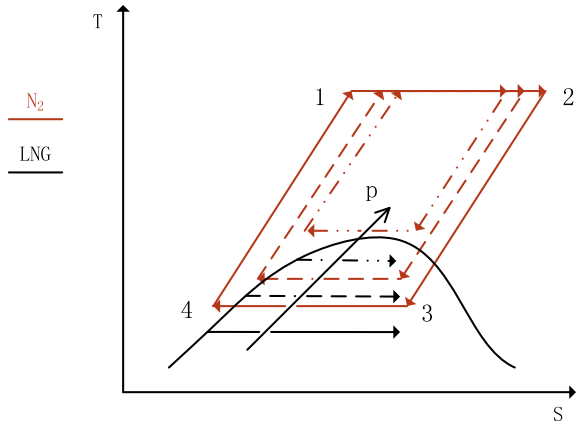

Fig. 8. The relationship between net work and vaporization pressure.

Obviously, when $\overline{T_H}/\overline{T_L} > 2$, $W_{\text{net}} > Q_{\text{out}}$, which shows the work output is larger than the heat released to the heat sink at the mean exothermic temperature. In other words, if the mean endothermic temperature of LNG is lower than a specific value, the work output of the basic process could be greater than the heat absorbed by LNG.

Similarly, when $\overline{T_H}/\overline{T_L} < 2$, $W_{\text{net}} < Q_{\text{out}}$ and when $\overline{T_H}/\overline{T_L} = 2$, $W_{\text{net}} = Q_{\text{out}}$.

For heat exchanger HX3, the heat released from the imaginary Carnot engine is absorbed by LNG. At the cross point of the exergy loss and the heat load curves in Fig. 11, we calculate that $Q_{\text{out}} = 264.15$ kW, $\overline{T_H} = 293.12$ K, $\overline{T_L} = 147.98$ K, then $\overline{T_H}/\overline{T_L} = 1.98 \approx 2$, $W_{\text{net}} = Q_{\text{out}}$. Actually, the heat exchanger HX3 does not have any work output. Therefore, W_{net} is the exergy loss of HX3. When the vaporization pressure is below 3.36 MPa, W_{net} will be greater than Q_{out}, so the exergy loss is more than the thermal load. Oppositely, when the vaporization pressure is above 3.36 MPa, W_{net} will be less than Q_{out}, so the exergy loss is lower than the thermal load.

For the heat exchanger HX2, there is no cross point between the thermal load curve and the exergy loss curve. Because under the working condition of HX2, $\overline{T_H}/\overline{T_L} < 2$ is always exact.

4.3. On the comprehensive efficiency of cold energy utilization

The comprehensive efficiency of cold energy utilization, η_{ce} is defined as the ratio of physical exergy gained in system $Ex_{\text{gain,SYS}}$ to the physical exergy provided by LNG $Ex_{\text{pay,LNG}}$, as shown in Eq. (15).

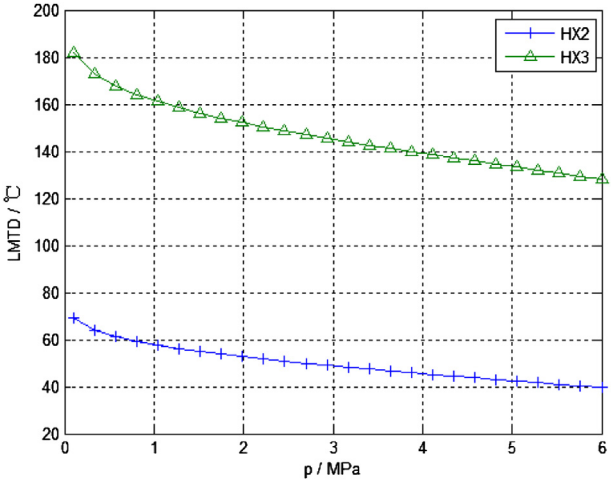

Fig. 9. Log mean temperature difference of HX2 and HX3 against vaporization pressure.

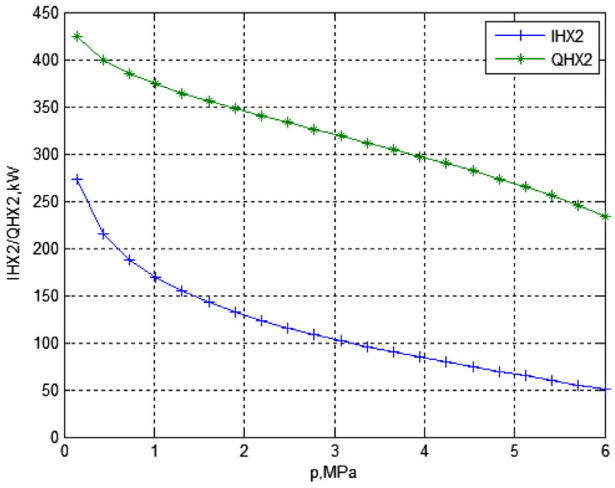

Fig. 10. The exergy loss and heat load against vaporization pressure in heat exchanger HX2.

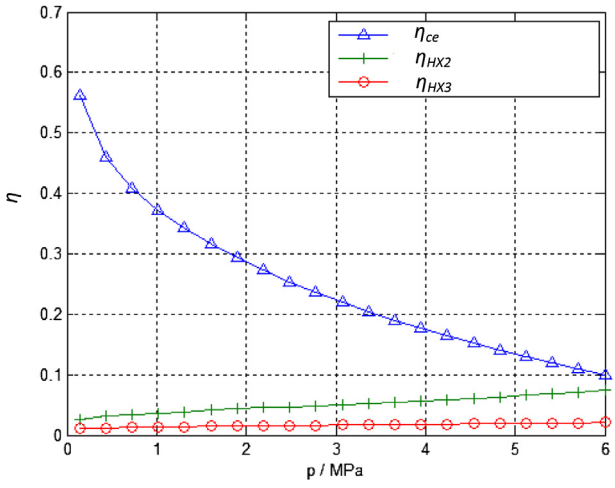

Fig. 12. The effect of vaporization pressure on comprehensive and heat exchanger efficiency of cold energy.

$$\eta_{ce} = \frac{Ex_{gain,SYS}}{Ex_{pay,LNG}} \tag{15}$$

where the physical exergy gained from the system comprises mechanical exergy and the cryogenic exergy transformed to other working fluids.

Under different vaporization pressures, the cold energy released in the LNG regasification process also vary. The effect of vaporization pressure on comprehensive efficiency of cold energy utilization is shown in Fig. 12. It can be seen that with the increase of vaporization pressure, the efficiency goes down rapidly because the net work of the Stirling cycle decreases as the vaporization increases as shown in Fig. 8. As vaporization pressure is 3 MPa, η_{ce} is equal to 0.25. Although the exergy efficiency of HX2 and HX3 increase with vaporization pressure, they are no more than 0.1. That is because the LMTD variation of these two heat exchanger impacted by vaporization pressure is tiny.

4.4. Effect of ambient temperature on net work

The effect of heat source temperature on net work under different vaporization pressures (1.0 MPa, 2.0 MPa and 3.0 MPa) is shown in Fig. 13. Here the heat source refers to the ambient temperature, which ranges from 0 °C to 30 °C. As seen from the graph, when the ambient temperature increases, the net work of the basic process increases as well for all three different vaporization pressures. If the heat source was replaced by something with higher temperature than ambient, such as waste water from power station, exhaust gas from turbine and even solar energy, the work output of basic process could be greatly increased.

4.5. Feasibility in improvements of the basic process

According to the thermodynamic analysis of the basic process, the heat exchanger performance could be optimized to improve the basic process. An improved process of LNG cryogenic generation with Stirling cycle combined with air liquefaction is proposed. The scheme of the improved cycle is shown in Fig. 14.

In the improved process, the heat exchanger HX2 and HX3 are used to precool processing air instead of cooling seawater. The processing air goes through HX2 and HX3 successively. The air liquefaction process in the improved process is based on the classic method

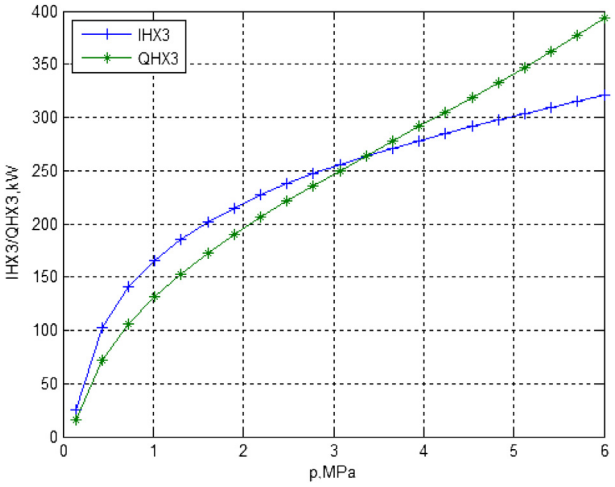

Fig. 11. The exergy loss and heat load against vaporization pressure in HX3.

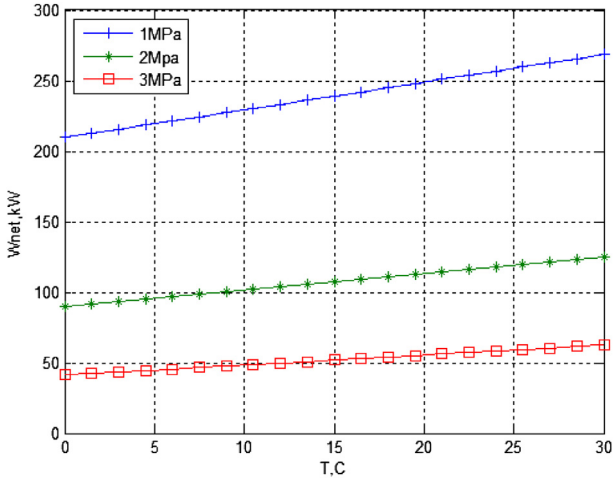

Fig. 13. The effect of ambient temperature to net work under different vaporization pressure (1 MPa, 2 MPa, 3 MPa).

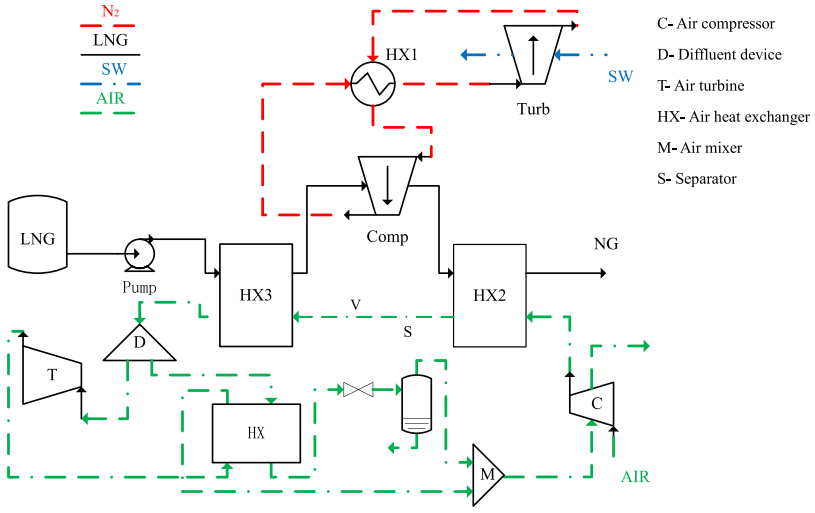

Fig. 14. The scheme of LNG cryogenic generation combined with air liquefaction.

for air liquefaction with a low pressure cycle using a special high-efficiency expansion turbine developed by Pyotr Kapitsa [23] in 1939.

Reducing CO_2 (carbon dioxide) emission of conventional energy systems involving combustion of fossil fuels is very important for environment. Thus, the heat exchanger HX2 and HX3 may be used to cool down exhaust gas from turbine for capturing CO_2 [24]. The feasibility will be proved in future work.

5. Conclusions

This paper discusses the Stirling cycle with nitrogen applied for LNG cryogenic power generation. The thermodynamic process of the basic process is analyzed. The parameters of the regasification process and the power cycle, as well as their sensitivities are also studied. More experiments are needed to study the Stirling cycle further.

By calculation, when LNG flow rate is raised to be 60 t/h (vaporization pressure is 3.0 MPa), the net work output of the basic process is 1667 kW. Compared with Senboku Daini cryogenic power station in Japan, the power output of the basic process is increased by 14.97%. The increase of output power can be explained by a series of assumptions, such as isothermal compression, isothermal expansion, etc.

The main factors affecting the generating capacity and the cryogenic generation efficiency of the Stirling cycle method are LNG vaporization pressure, LNG mass flow rate and heat source temperature. To increase the performance of the basic process, a high temperature heat source like exhaust gas should be used, and LNG evaporation pressure and mass flow rate should be controlled in an acceptable range.

The largest exergy loss equipment in the basic process is the seawater heat exchanger. The exergy losses in heat exchangers account for 92.65% of the total exergy loss in the process. The possible way to reduce exergy loss in these heat exchangers is proposed in the improved process.

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Nomenclature

c_v: specific heat at constant volume J/(kg K)
EX: exergy rate kW
EX_{gain}: exergy gain rate
Ex_{pay}: exergy pay rate
$ex_{T,p}$: specific physical exergy rate
H: specific enthalpy kJ/kg
I: exergy loss rate kW
M: mass flow rate kg/s
P: pressure MPa
p_0: ambient pressure MPa
Q: specific heat kJ/kg
Q_c: cold energy rate kW
Q_{in}: heat input rate kW
Q_{out}: heat output rate kW
R_g: gas constant J/(kg·K)
s: specific entropy kJ/(kg·K)
T, t: temperature °C
T_0: ambient temperature °C
T_H: heat source temperature °C
$\overline{T_H}$: mean endothermic temperature °C
T_L: heat sink temperature °C
$\overline{T_L}$: mean exothermic temperature °C
w: specific work kJ/kg
W_{net}: net work output rate kW
W_g: power output rate kW
x: cryogenic power coefficient %
η_{ex}: exergy efficiency %
η_g: machinery conversion efficiency %

Subscripts

c: cold
ce: cold energy
g: generator
N_2: nitrogen
T: Isothermal process
v: Isochoric process
0: ambient state
$1…4$: states of nitrogen
$1L…5L$: states of liquid natural gas

Abbreviations

Comp: compressor
HX: heat exchanger
LMTD: log mean temperature difference
LNG: liquefied natural gas
NG: natural gas
SYS: system
SW: seawater
Turb: Turbine